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Energy Savings Potential of Passive Chilled Beam System as a Retrofit Option for Commercial Buildings in Different Climates

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ABSTRACT

This study considers a cooling system retrofit for a commercial building in different climates by introducing a passive chilled beam system. A typical single duct VAV system, widely used in large office buildings, was considered as a baseline scenario; a combination of the air system and passive chilled beams was considered as a retrofit scenario. The retrofit with minimum modifications in this study includes installations of (1) multiple passive chilled beam units, (2) additional pumps and a closed water loop for the passive chilled beam units and (3) a heat exchanger where the chilled beam's water loop exchanges heat with the return side of the chilled water loop. The results showed total energy savings up to 22% for climates where the sensible load is much higher than the latent load.

1. INTRODUCTION

Reducing energy consumption in buildings is an important part of reducing global energy usage and environmental impact. It has been reported by the International Energy Agency (IEA, Energy technology perspectives 2012) that the global energy demand in buildings will be doubled in 2050 due to the “rising number of residential and commercial buildings in response”. More than two-thirds of existing buildings are expected to remain until 2050. Therefore, significantly reducing energy use in the building sector requires retrofitting existing buildings with advanced technologies. Furthermore, modeling of these technologies is a critical part of assessing their economics towards realizing success in the marketplace.

Passive chilled beams can be considered passive systems, which rely only on radiation and natural convection for sensible heat transfer and do not employ active fans; while active systems include forced air movement with fan operation to enhance air flow and convective heat transfer. Typically, latent load requirements are handled separately using a dedicated outdoor air system (DOAS) that conditions only the ventilation air. Compared to conventional all-air systems, this can lead to reduced overall cooling requirements: a significant benefit of passive systems. Since the amount of air carried to the space can be reduced significantly, there is also less fan energy consumption compared to conventional systems. Furthermore, the reduction of air flow allows downsized ductwork. The cooling load can be further reduced by taking advantage of the improved comfort associated with radiant cooling of occupants, which can allow a somewhat higher zone temperature for the same comfort level. Furthermore, higher radiant surface temperatures (above the dew point of the air) lead to higher operating temperatures than those of conventional cooling systems, and improved energy efficiency for chillers. It is also possible to couple the radiant cooling system with energy storage within the building thermal mass, to allow load shifting and reductions in peak load.

Energy modeling tools are important for designing building systems and for assessing new technologies such as passive ceiling cooling systems. To evaluate both thermal comfort and energy saving aspects of passive chilled beams, the model should accurately take into account the primary heat transfer mechanisms of radiation and natural convection which are strongly coupled to the specific characteristics and conditions within the spaces where they are

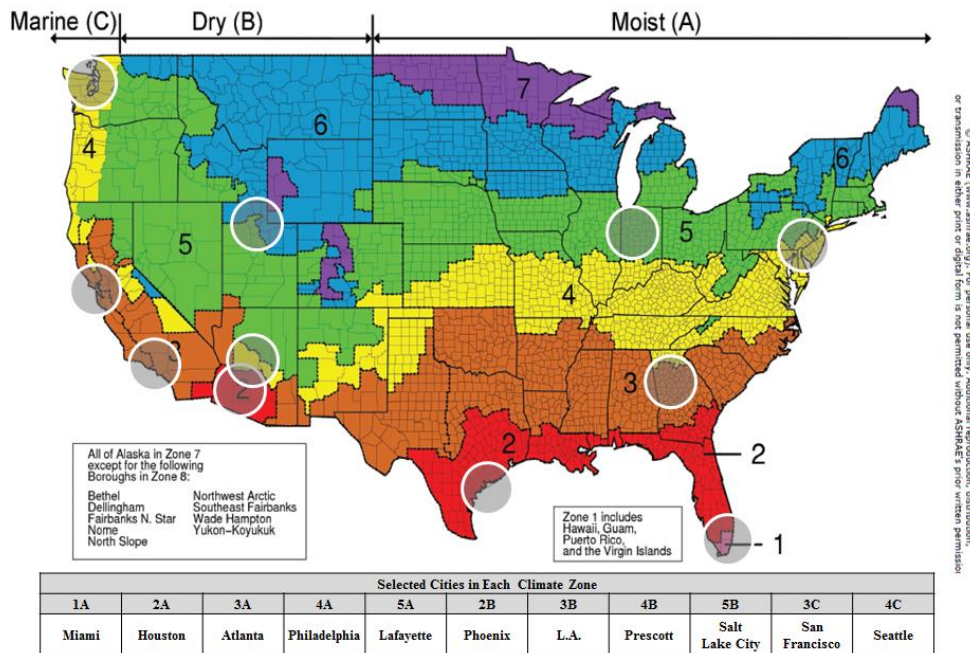


Figure 1: Eleven cities representing the different climate zones

installed. However, it is also possible to partially analyze the aspect of decoupling sensible and latent loads for passive chilled beam applications with simplified models. Since the capital cost of passive chilled beam systems is the major barrier of introducing the system into the market place, this study focuses on passive chilled beams as a retrofit option with minimum modification of the original cooling system. The study considers energy savings associated with decoupling of sensible and latent loads for the same zone temperature setpoints, but does not consider benefits of improved thermal comfort that may result.

2. BUILDING ENERGY MODEL DESCRIPTION

The details and parameters of the simulation model are described in the following sections. The model was implemented in TRNSYS software (TRNSYS, 2004).

2.1 Climatic data

Different climatic zones were considered that have different ambient humidity levels (marine, dry and moist) and five different latitudes (zone 1 to 5) based on the U.S. climate zone map in ANSI/ASHRAE Standard 90.1 (ASHRAE, 2013). Based on this classification, eleven different cities were selected (Figure 1) to import TMY 2 weather data into the simulation. Since the study focuses on the cooling system's performance, zone 6 and higher were not considered. Detailed definitions of each climate zone can be found in ANSI/ASHRAE Standard 90.1 (ASHRAE, 2013).

2.2 Building Envelope Parameters

The building envelope model was developed based on construction drawings of one of the new Living Lab offices in the Ray W. Herrick Laboratories at Purdue University (West Lafayette, Indiana) as shown in Figure 2. Table 1 includes details of the envelope components. Although the actual office is equipped with a double façade, the model is applied with a regular double-glazed window to focus more on the relative performance of the two cooling system alternatives. The office has a 117 m² floor area with the height of 4.6 m. North, east and bottom side of the space is adjacent with other spaces in the building and south and west wall are exposed to the ambient. The window to wall ratio is about 50% of the South wall.

2.3 Internal Gains

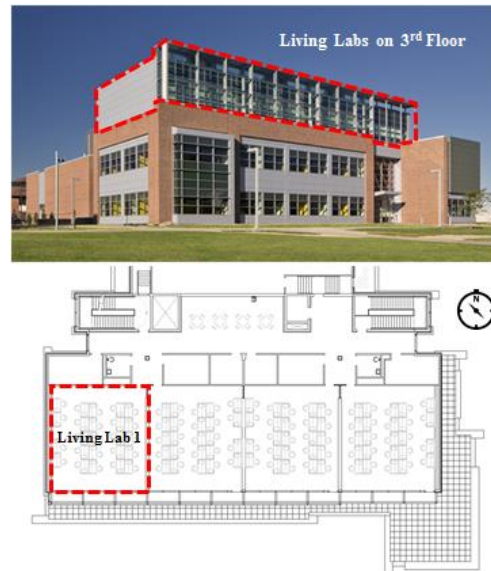


Figure 2: Exterior view and plan of the studied office space

Internal gains schedules for occupancy, lighting and plug loads were adopted from ASHRAE Standard 90.1 (ASHRAE, 2010) as shown in Figure 3. These schedules represent typical load profiles of commercial building.

2.4 Conventional Air System

A typical single duct VAV system, which is widely used in large office buildings (Thornton et al., 2011), was considered as the cooling system for the baseline scenario. As shown in Figure 4 (a), the system includes a cooling coil, supply and return fan and an economizer. The preheating coil, reheating coil and VAV box were not included in this study to simplify the approach for the consideration of a summer cooling season. The chilled water source for the cooling coil is provided from a water cooled chiller which is driven by electricity and a variable speed pump was used to provide chilled water from the chiller to the cooling coil.

The control sequence of the air system is basically to meet the space sensible load requirement while latent load is mostly covered in the same cooling process at the cooling coil. The speed of the supply fan (Fan 1) is controlled to meet the room temperature setpoint (23.9°C) and the speed of the pump (Pump 1) at chilled water loop is controlled to meet the cooling coil outlet air temperature setpoint (13°C). The return fan (Fan 2) is assumed to be synchronized with the supply fan's operation. The chiller is controlled to provide 7°C outlet temperature whenever the cooling signal is on. The economizer is controlled based on temperature difference between room setpoint and outdoor air.

Table 1: Specification of Envelope Components

Type	Layer Description	Thickness	U-Value
		m	W/m ² K
Roof	Insulation	0.1	0.221
	Concrete	0.038	
Raised Floor	Concrete	0.3	0.086
	Air	0.102	
External Wall	Gypsum Board	0.013	0.358
	Insulation	0.105	
	Stucco	0.025	
Internal Wall	Gypsum Board	0.016	0.136
	Air	0.184	
	Gypsum Board	0.016	
Window	Double pane, Low-e, IGU	-	1.43

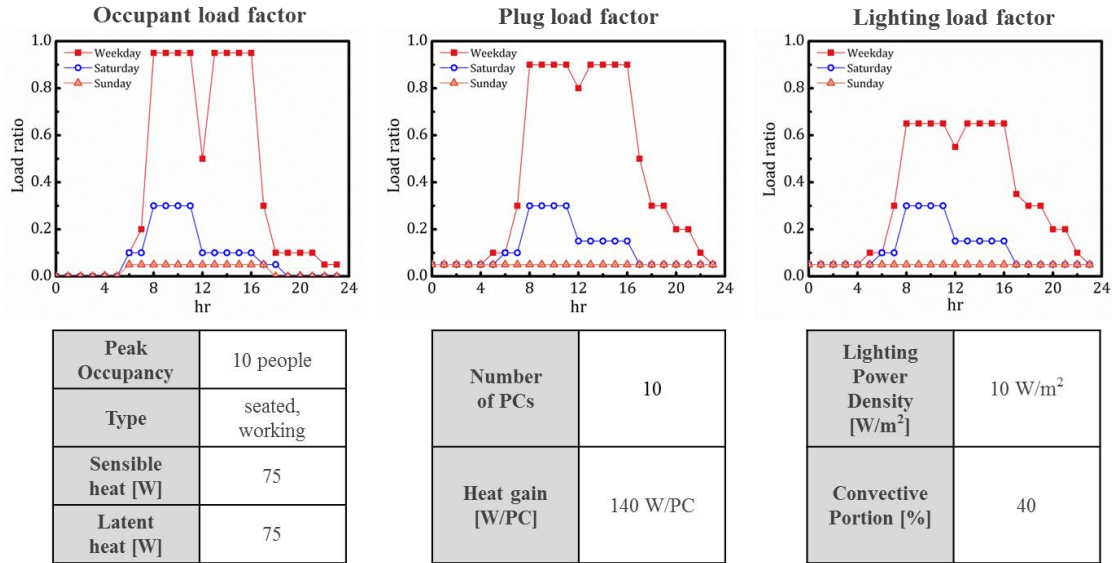


Figure 3: Specification of internal gains

The minimum outdoor air requirement (0.4 ACH) is applied to the system based on ANSI/ASHRAE Standard 62.1 (2010), thus even when outdoor air is not favorable, the economizer still includes the minimum outdoor air to the supply air stream. A thermostat was used to control the cooling signal with $\pm 1^\circ\text{C}$ deadband from the room temperature setpoint and a night setback schedule was applied besides the occupied hours (8 a.m. to 5 p.m.) for the room temperature setpoint ($+4^\circ\text{C}$).

2.5 Passive Chilled Beam System

In this case, the same air system is used for dehumidification and multiple passive chilled beams, connected to a separate water loop as shown in Figure 4 (b), are employed to satisfy the main sensible cooling load. The separate water loop for the chilled beams is coupled with the air system's cooling coil by a heat exchanger. The use of the heat exchanger allows variation of the pump speed (Pump 3) at the passive chilled beam's water loop to meet the sensible load requirement. The air system's chilled water loop also utilizes variable speed capability to meet the latent load requirement. As shown in the Figure 4 (b), two additional water pumps (Pump 2 and 3) are included over the conventional air system's configuration. The additional water pump and a bypass loop were added prior to the cooling coil to allow sufficient chilled beam cooling capacity. However, the temperature inlet to the passive chilled beams is controlled to avoid condensation. The cooled water was provided to each passive chilled beam unit individually, providing the same inlet temperature to each unit, as it is usually applied in practical design.

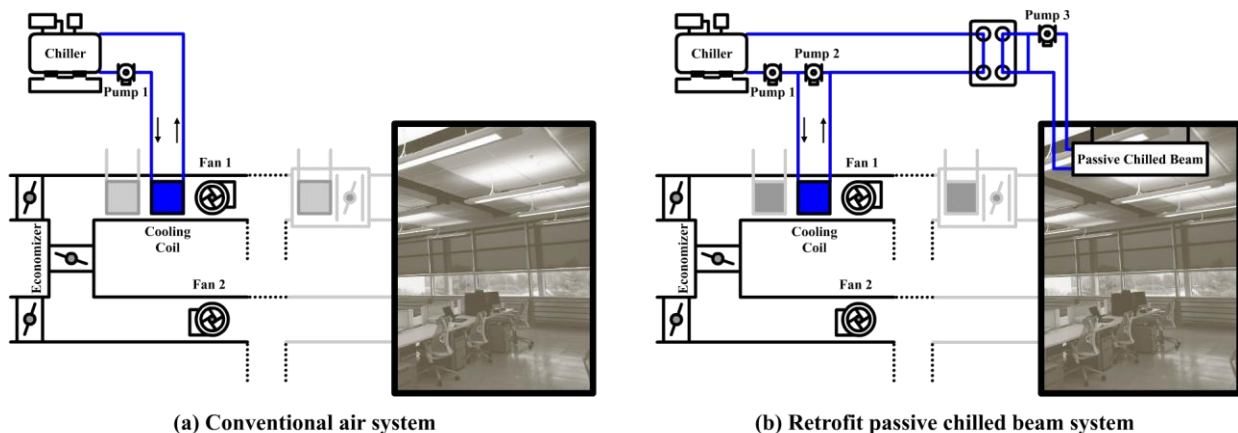


Figure 4: HVAC system layout of conventional and retrofit scenarios

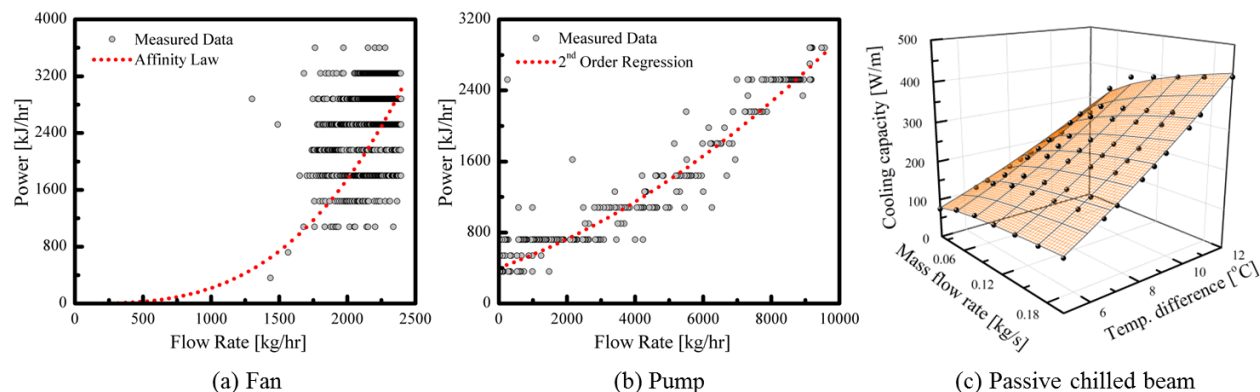


Figure 5: Performance characteristics of fan, pump and passive chilled beam

The control sequence of the passive chilled beam system is to meet the latent load requirement with the air system while the passive chilled beam system covers the remaining sensible load. While the chiller and the pump (Pump 1) within the chilled water loop are controlled with the same sequence as in the conventional air system's scenario, the speed of the supply fan (Fan 1) is controlled to meet the humidity setpoint ($0.01 \text{ kg}_{\text{water}}/\text{kg}_{\text{air}}$) in the space. One additional pump (Pump 2) in the cooling coil's bypass loop is controlled to provide 9°C water temperature to the heat exchanger whenever there is a cooling requirement at the passive chilled beam. While the air system is controlled based on the latent load requirement, the coil cooling rate will generally not be sufficient to maintain the temperature of the space at the setpoint of 23.9°C during summer. Thus, the passive chilled beams' output is controlled to meet the temperature setpoint. Nominal water temperature (13°C) and mass flow rate (0.1 kg/s) were used for the passive chilled beams' operation, while the number of passive chilled beam units was used as a variable to calculate the maximum required number of passive chilled beams needed in the space. A humidistat was used to control the cooling signal with $\pm 0.001 \text{ kg}_{\text{water}}/\text{kg}_{\text{air}}$ deadband from the room humidity ratio setpoint and a night setback schedule was applied besides the occupied hours (8 a.m. to 5 p.m.) for the room humidity ratio setpoint ($+0.003 \text{ kg}_{\text{water}}/\text{kg}_{\text{air}}$).

2.6 HVAC Components

Water cooled chiller, fan, pump and passive chilled beam models are incorporated in the building energy model. The performance of these components needs to be estimated to correctly evaluate the effect of a retrofit scenario. The water cooled chiller's performance was predicted with one of the models from the TRNSYS library (Type 666). A normalized performance map was used to calculate the power of the chiller from different conditions of rated capacity, rated COP, water mass flow rate, chilled water leaving temperature and cooling water inlet temperature. Performance of the fan and pump were applied based on actual measurements. Supply flow rate and power consumption of the fan and the pump were directly coupled by using the affinity laws for the fan and a 2nd order polynomial for the pump as shown in Figure 5 (a) and (b). A regression model available for the passive chilled beam (based on the experimental measurements) was used (DOE 2.1E Manual, 2013). The surface fit shown in Figure 5 (c) was used to calculate the heat extraction rate of the passive chilled beam based on water mass flow rate and temperature difference between indoor air temperature and water inlet temperature.

3. RESULTS

The components described in the previous section were incorporated in TRNSYS to conduct transient simulations of both the conventional air system scenario and the retrofit passive chilled beam scenario for the cooling season (July and August) in all 11 climatic zones. A 15-min time step was used in the simulation. Total energy consumption per unit floor area for all cases are shown in Figure 6. The most significant part of the savings comes from the fan's reduced energy consumption due to reduced coil loading. In every climate, more than 70% of the fan energy was saved in the retrofitted scenario. The required supply air flow is significantly less because the coil meets a much smaller load when controlled based on latent requirements rather than for sensible load. Furthermore, fan power depends on the cube of the air flow leading to substantial savings. As shown in Figure 7, the fan energy consumption is driven by the sensible load profile in the conventional system, while in passive chilled beam

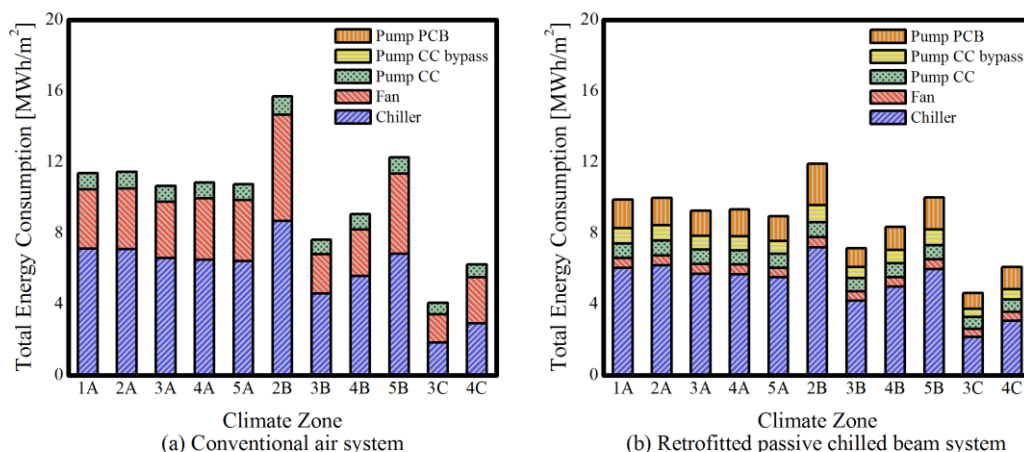


Figure 6: Total energy consumption in different climatic zones

configuration it is driven by the latent load. Since the latent load is smaller than the sensible load in every climate considered in this study, fan energy can always be saved.

As shown in Figure 6, chiller energy can also be saved in climatic zones A and B with savings ranging from 9 to 17%. However, there are insignificant savings in marine zones (C). The chiller savings are the result of decoupling of the sensible and the latent load. Since the conventional air system is operated based on the sensible load only, dehumidification is driven by the profile of the sensible load, which, in certain climates, might provide excess dehumidification to the space. Figure 8 (a) shows the excess dehumidification (latent load on cooling coil) for the conventional scenario compared to Figure 8 (b) for the retrofitted scenario. It also shows that almost the same magnitude of the sensible load is met by the cooling coil in the conventional scenario and by the cooling coil and the passive chilled beam in the retrofitted scenario. Thus, since the chiller's power is driven by both sensible and latent loads as depicted in Figure 7, the total load on the chiller which is the total magnitude of the bars in Figure 8 is reduced in most climatic zones in the retrofitted scenario.

The total energy savings in different climatic zones (calculated from Figure 6) are shown in Figure 9. Variable energy savings are due to the difference between the magnitude of the sensible and latent loads. Compared to the humid climates (A), dry regions (B) will have less latent load but they might have either higher sensible load (2B and 5B) or lower sensible load (3B and 4B). Marine regions (C) have both less sensible and latent load compared to humid regions. Thus, compared to humid regions, climatic zone 2B and 5B will have higher energy savings since excess dehumidification will increase according to higher sensible load, while the actual latent load will decrease in dryer regions. The savings due to excess dehumidification are depicted in Figure 10: in the conventional scenario, the cooling system will be driven by the profile of the sensible load. If the actual latent load is less than the

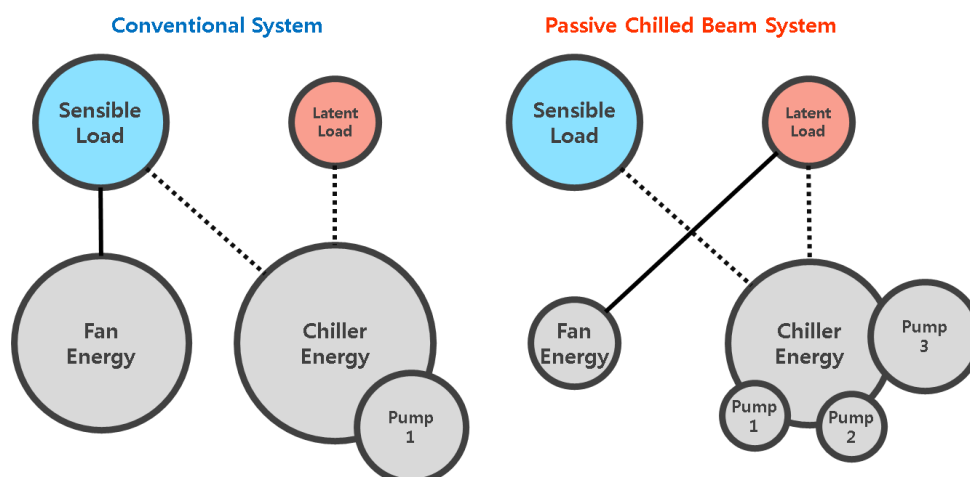


Figure 7: Driving force of each component in both scenarios

dehumidification provided by the cooling system, the difference between excess dehumidification and the actual latent load, shown in Figure 10 (b), can be saved in the retrofitted scenario since the load is decoupled and met separately by the air system and passive chilled beam system. Thus, it will be favorable to apply the passive chilled beam system in regions where the sensible load is high and the latent load is low.

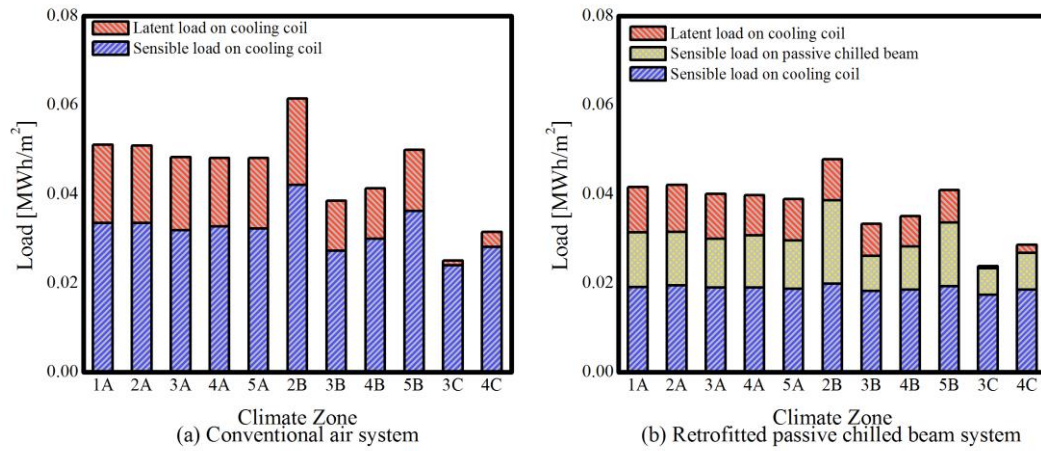


Figure 8: Load on each component

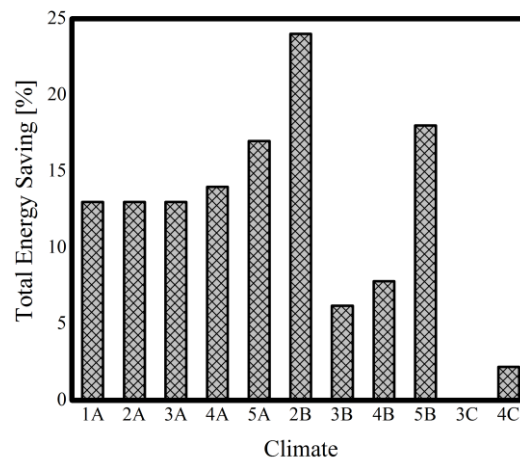


Figure 9: Total energy saving in each climate

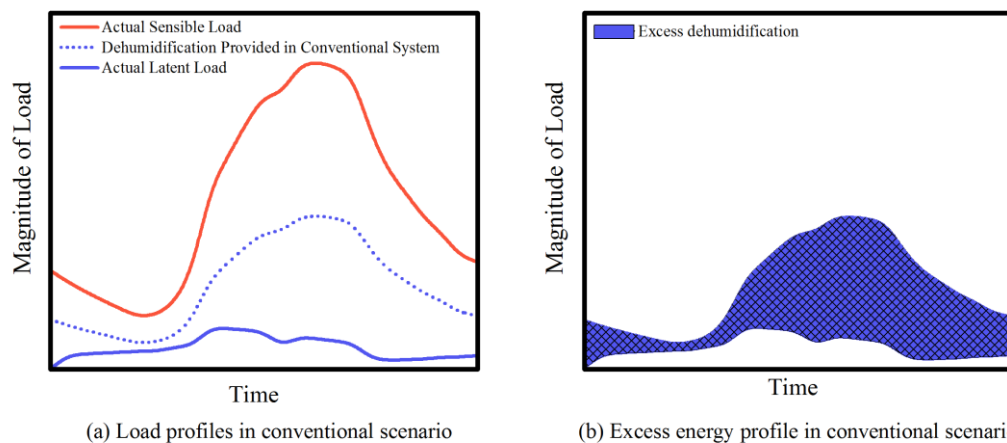


Figure 10: Excess dehumidification in conventional air system which can be saved with the chilled beam configuration.

4. CONCLUSIONS

The following conclusions were found in considering a passive chilled beam system as a retrofit option:

- Total energy savings between 8 to 24% in humid and dry climate zones.
- Passive chilled beams as a retrofit option was not favorable in marine regions.
- Substantial fan energy can be saved in every climate (~70%) depending on the magnitude of the latent load.
- Chiller energy can be saved in both humid and dry regions.

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